

ANALYSIS OF ENERGY RECOVERY VENTILATOR SAVINGS FOR TEXAS BUILDINGS

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ABSTRACT

This analysis¹ was conducted to identify the energy cost savings from retrofitting Texas buildings with air-to-air ERV (Energy Recovery Ventilator) systems. This analysis applied ERV and psychrometric equations in a bin-type procedure to determine the energy and costs required to condition outside air to return-air conditions. This analysis does not consider interactions with the air-handling system; therefore the effects of economizers, reheat schemes, variable flow rates and other adaptive components were not considered.

This analysis demonstrates that ERV cost-effectiveness is largely dependent upon the building location in Texas (i.e., climate conditions) and outside air fraction:

- For a typical laboratory building that requires 100% outside air, an ERV could save roughly \$1.00 to \$1.50 per cubic foot per minute (CFM) of outside air during a one year period.
- For a typical office building that only requires 10% outside air, an ERV could save up to \$1.00 per CFM of outside air over the period of one year.

INTRODUCTION

Motivation

Commercial buildings routinely exchange indoor air with outside air in order to remove indoor air contaminants. Typical laboratory buildings may require 100% of the supply air to be taken from the outside, whereas typical office buildings may require only 10% outside air. Conditioning this outside air often requires a significant amount of energy in climates such as Texas (ASHRAE Fundamentals 2005). Therefore there is an opportunity to save energy by transferring heat and moisture between the outside air and the exhaust air. Energy Recovery Ventilators (ERVs) are devices that attempt to save energy by using exhaust air to economically condition outside air to less extreme conditions.

ASHRAE Standard 90.1-2007 is the current building code for State Agencies. For other commercial buildings it is the 2000 IRC as modified by the 2001 Supplement, which refers to ASHRAE 90.1-1999. Both the 90.1-1999 and the 90.1-2007 standards require that all large air handling units (5,000 CFM or more) with 70% or more outside air are required to install an exhaust air energy recovery device. Thus the building code requires installation of energy recovery devices in selected new buildings (and renovations brought to current code). For existing buildings, however, it may not be obvious where it is cost-effective to retrofit buildings with ERV technology.

ERV Mechanics

By definition, heat transfer between two air flows can only occur if there is a temperature difference (Incropera & DeWitt 1996). Similarly, mass transfer between two regions can only occur if there is a pressure difference and a pathway (ASHRAE Systems 2004). ERVs exploit such heat and mass transfer mechanisms to economically pre-condition outside air to less extreme conditions. While ERVs use various methods to transfer sensible and latent heat from one air stream to another, in this analysis we generalized the ERV as shown in Figure 1.

A key parameter that describes how well ERVs transfer sensible and latent heat from one air stream to another is the ERV effectiveness ε . ASHRAE Standard 84-1991 defines effectiveness as (ASHRAE Systems 2004)

$$\varepsilon = \frac{\text{Actual transfer of moisture or energy}}{\text{Maximum possible transfer}}. \quad (1)$$

ERV Psychrometric Path

Figure 2 shows an ERV operating in the *Summer* in San Antonio with outside air conditions of 92.5°F dry-bulb and 73°F mean coincident wet-bulb temperatures and in the *Winter* in San Antonio with outside air conditions of 42.5°F dry-bulb and 39°F mean coincident wet-bulb temperatures. The ERV assumptions and specifications are detailed in Appendix A.

¹ This paper is based on an internal technical report written by the same authors and listed in the references.

Mixed Air condition is much closer to the Return Air condition (#3).

METHODOLOGY

This analysis used the psychrometric equations provided in the *2005 ASHRAE Fundamentals Handbook* and ERV equations in the *2004 ASHRAE HVAC Systems and Equipment Handbook*, in a bin-type analysis to determine the energy and costs needed to condition outside air to return-air conditions. This analysis did not consider interactions with the air-handling system; therefore the effects of economizers, reheat schemes, variable flow rates and other adaptive components were not considered in this preliminary analysis.

The assumptions² used in this analysis include, but are not limited to:

- Outside air provided 24 hours per day throughout the year.
- Return air conditions are constant year-round at 75°F and 50% relative humidity.
- An ERV that provides 25,000 CFM of outside air with a sensible and latent effectiveness assumed to be 64% and 60%, respectively.

Scenario Summaries

Unfortunately, operating an ERV under certain conditions can actually increase the overall HVAC energy use. For example, when Outside Air conditions are close to the AHU's coil leaving condition it may not be desirable to pre-condition the incoming Outside Air with an ERV. In order to guide this simplified analysis, it is useful to consider scenarios for operating the ERV, summarized in Table 1 below.

Table 1: Summary of the ERV Scenarios.

Scenario:	ERV control logic:	ERV Energy Savings:
Scenario #1	ERV is always on for all temperature bins.	Upper-bound estimate, because it includes heating "savings" when Outside Air conditions are moderate (e.g. 66°F).
Scenario #2	ERV is on if $T_{OA} > T_{RA}$. See footnote 3.	Lower-bound estimate, since heating savings is not considered.

² See Appendix A for a complete list of assumptions and parameter values.

Scenario #3	ERV is on if $T_{OA} > T_{RA}$ or $T_{MA, bypass\ open} < T_{ERV\ cutoff}$.	Compromise between upper and lower-bound estimates. <i>The current analysis used this scenario exclusively.</i>
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Scenario #1 models an ERV that runs 100% of the time, no matter the outside conditions.

Unfortunately, the energy saved when the ERV is operating as a heating device (where $T_{OA} < T_{RA}$) may be over-estimated. For example, when the outside temperature is 66°F it is unlikely that the outside air is heated to a Return Air condition of 75°F. Therefore the energy savings predicted in this scenario represent an upper-bound estimate.

Scenario #2 removes temperature bins where the ERV heats the Outside Air, resulting in an ERV that operates solely to cool outside air. When the ERV is off, an ERV bypass opens to supply the same amount of fresh air directly to the system. Since the cooling savings are real and this scenario ignores any heating savings, the energy savings predicted in this scenario are considered a lower-bound estimate.

Scenario #3 is an attempt to determine the savings for an ERV with a somewhat intelligent bypass. This scenario considers all cooling situations ($T_{OA} > T_{RA}$) and situations where heating of the Mixed Air is likely ($T_{MA, Bypass\ open} < T_{ERV\ cutoff}$).

$T_{MA, Bypass\ open}$ (the temperature of the Mixed Air with the bypass open) is defined in Eq. (45) in Appendix B. $T_{ERV\ cutoff}$ (the ERV Cutoff Temperature) is a threshold used to help determine if the ERV is bypassed or not.⁴ This scenario is a compromise between the other two scenarios: it attempts to ignore the band of bin temperatures where the ERV could claim non-existent heating savings (e.g., Outside Air temperature of 66°F).

This analysis compares the amount of energy needed to condition Outside Air to Return Air conditions, with and without an ERV. While such a blackbox approach is simple, it ignores the effects from the overall air-handling system (AHU schemes and setpoints, constant-volume vs. variable-volume airflow, single-duct vs. dual-duct air distribution methods, reheat schemes, and etc.).

³ In all our calculations, we set the Return Air temperature $T_{RA} = 75^\circ\text{F}$.

⁴ In all our calculations, we set the ERV Cutoff Temperature $T_{ERV\ Cutoff} = 55^\circ\text{F}$.

Finally, regardless the scenario, this analysis considers the ERV to be either on or off; there is no modulation of airflow. When the ERV is off, an ERV bypass is fully opened to supply the same amount of fresh air directly to the system, and air flow through the ERV is shut-off.

Calculating Preconditioning Energy in Temperature Bins

The preconditioning energy required to condition Outside Air to Return Air conditions is calculated with and without an ERV.

Without an ERV, the sensible and latent energy rate (in Btu/hr) that would be required to condition the Outside Air to the Return Air condition is calculated as (ASHRAE Fundamentals 2005):

$$\dot{q}_{sensible, no\ ERV} = C_s \dot{V}_{OA} (T_{OA} - T_{RA}) \quad (2)$$

$$\dot{q}_{latent, no\ ERV} = C_l \dot{V}_{OA} (W_{OA} - W_{RA}) \quad (3)$$

where

- heat transfer rates are in Btu/hr units,
- C_s is the air sensible heat factor in Btu/hr·°F·CFM (typically, $C_s = 1.1$),
- C_l is the air latent heat factor in Btu/hr·CFM (typically, $C_l = 4840$),
- Outside Airflow \dot{V}_{OA} is in CFM,
- temperatures T are in Fahrenheit, and
- humidity ratios W are in lb. moisture/lb. dry-air.

With an ERV, the sensible and latent energy rate (in Btu/hr) that would be required to condition the Fresh Air supplied by the ERV to the Return Air condition is

$$\dot{q}_{sensible, ERV} = C_s \dot{V}_{OA} (T_{FA} - T_{RA}) \quad (4)$$

$$\dot{q}_{latent, ERV} = C_l \dot{V}_{OA} (W_{FA} - W_{RA}) \quad (5)$$

Next the number of hours n the ERV is on for each temperature bin is determined. Scenario #3 (see Table 1) was used to determine whether or not a temperature bin had the ERV on or off. If the ERV was on, then n is the number of hours in the original weather bin; if the ERV was off then this value is zero.

The annual energy required for each bin was calculated by multiplying the number of hours n with the respective heat transfer rates for that bin (in Eq. (2) through (5)):

$$q_{sensible, no\ ERV} = \frac{n \dot{q}_{sensible, no\ ERV}}{1,000,000} \quad (6)$$

$$q_{latent, no\ ERV} = \frac{n \dot{q}_{latent, no\ ERV}}{1,000,000} \quad (7)$$

$$q_{sensible, ERV} = \frac{n \dot{q}_{sensible, ERV}}{1,000,000} \quad (8)$$

$$q_{latent, ERV} = \frac{n \dot{q}_{latent, ERV}}{1,000,000} \quad (9)$$

where the bin time n is measured in hours, heat transfer rates \dot{q} are in Btu/hr units, and energy q is in MMBtu.

Calculating Annual Preconditioning Energy

Without an ERV.

The summation of all the bins of $q_{sensible, no\ ERV}$ (Eq. (6)) that are *greater* than zero gives the Cooling Sensible Energy without an ERV

$E_{sensible\ cooling, no\ ERV}$ in Eq. (10). Essentially this is the annual energy required to sensibly cool Outside Air to Return Air conditions. The summation of the same bins that are *less* than zero gives the Heating Sensible Energy without an ERV

$E_{sensible\ heating, no\ ERV}$ in Eq. (11). The index i reminds us that these quantities are arrays; one value for each temperature bin. The positive and negative exponents symbolize all the positive and negative bin values, respectively.

$$E_{sensible\ cooling, no\ ERV} = \sum_i (q_{sensible, no\ ERV, (i)})^+ \quad (10)$$

$$E_{sensible\ heating, no\ ERV} = \sum_i (q_{sensible, no\ ERV, (i)})^- \quad (11)$$

Similarly, the summation of all bins of $q_{latent, no\ ERV}$ (Eq. (7)) that are *greater* than zero gives the Cooling Latent Energy without an ERV $E_{latent\ cooling, no\ ERV}$ in Eq. (12); essentially this is the annual energy required to dehumidify Outside Air to Return Air conditions. The summation of the same bins that are *less* than zero gives the Heating Latent Energy without ERV $E_{latent\ heating, no\ ERV}$ in Eq. (13).

$$E_{latent \text{ cooling, no ERV}} = \sum_i (q_{latent, no ERV, (i)})^+ \quad (12)$$

$$E_{latent \text{ heating, no ERV}} = \sum_i (q_{latent, no ERV, (i)})^- \quad (13)$$

With an ERV.

Next the energy and costs required to condition ERV-supplied Fresh Air to Return Air conditions are determined. The “with ERV” energy is determined the same way as “without ERV” except using $q_{sensible, ERV}$ and $q_{latent, ERV}$ (Eq. (8) and (9)) instead of $q_{sensible, no ERV}$ and $q_{latent, no ERV}$.

$$E_{sensible \text{ cooling, ERV}} = \sum_i (q_{sensible, ERV, (i)})^+ \quad (14)$$

$$E_{sensible \text{ heating, ERV}} = \sum_i (q_{sensible, ERV, (i)})^- \quad (15)$$

$$E_{latent \text{ cooling, ERV}} = \sum_i (q_{latent, ERV, (i)})^+ \quad (16)$$

$$E_{latent \text{ heating, ERV}} = \sum_i (q_{latent, ERV, (i)})^- \quad (17)$$

The ERV fan energy use is defined as

$$E_{ERV \text{ fan}} = P_{fan, Total} \sum_i n_{(i)} \quad (18)$$

where $P_{fan, Total}$ is the total ERV fan power (see Eq. (31)), n is the number of hours the ERV is on per temperature bin, and index i cycles through all available bins.

Calculating Annual Savings

Component energy costs for conditioning Outside Air to Return Air conditions were determined by multiplying each energy component (Sensible Cooling, Sensible Heating, Latent Cooling, Latent Heating, and ERV Fan Energy) with a specified energy unit cost for both “without ERV” and “with ERV” operation.

Energy savings and energy cost savings for each component (Sensible Cooling, Sensible Heating, Latent Cooling, Latent Heating, and ERV Fan Energy) were determined by subtracting the

respective energy values for “with ERV” from “without ERV” values.⁵

Percent savings are determined by dividing the cost savings with the respective “without ERV” cost. For example, the percent saved in Sensible Cooling is determined by⁶

$$\text{Percent Saved in Sensible Cooling} = \frac{\text{Costs saved in Sensible Cooling}}{\text{Cost of Sensible Cooling Without ERV}} \quad (19)$$

Additionally, the cost savings per CFM of outside air (in \$/CFM_{OA} units) is calculated. This normalized metric is useful when investigating the cost effectiveness of ERVs.

Total energy cost savings are determined by summing four of the component energy cost savings: Sensible Cooling, Latent Cooling, Sensible Heating, and ERV Fan Energy. Energy cost savings from ERV Latent Heating are not considered in this analysis.

It is useful to consider the payback time required to collect enough cost savings to offset the initial expense in acquiring the ERV. If we ignore interest, inflation, tax and tax incentives, installation/retrofit costs, and maintenance costs the simple payback time (in years) for the ERV can be determined by

$$\text{Payback Time} = \frac{\text{Initial Cost}}{\text{Annual Cost Savings}} \quad (20)$$

While Eq. (20) is fairly simple, ignoring the effect of the other costs is not necessarily prudent or recommended. For example, anecdotal evidence from the Austin Independent School District suggests that ERV maintenance costs are significant. In addition, retrofit installation costs could be significant if the exhaust air is not routed to a central location where it can be utilized by the ERV. Finally, interest and inflation “costs” may become significant, especially for long payback periods.

⁵ Cost savings for the ERV Fan Energy is negative because the ERV Fan Energy for the “without ERV” condition is zero.

⁶ Percent savings for the ERV Fan Energy is not applicable because the denominator is zero.

ANALYSIS SUMMARY

Results

Multiple scenarios were performed using the previously-described analysis method. The independent variables were Outside Air Fraction and weather data for five Texas cities (Brownsville, San Antonio, Dallas, Amarillo, and El Paso). The dependent variable of interest was the normalized total energy cost savings (Saved $\$/\text{CFM}_{\text{OA}}$ per year) using Scenario #3 (see Table 1). Note that savings from Latent Heating with the ERV are not included, since it is assumed that building humidification is not available.

Figure 3 shows the normalized savings (Saved $\$/\text{CFM}_{\text{OA}}$ per year) for selected Texas cities and buildings operated at various Outside Air Fractions. Office-type buildings may typically have a 10% Outside Air fraction; laboratories often require a 100% Outside Air fraction. Note that the energy cost savings at 50% Outside Air fraction is not necessarily near the average of the energy cost savings at 10% and 100% Outside Air fractions.

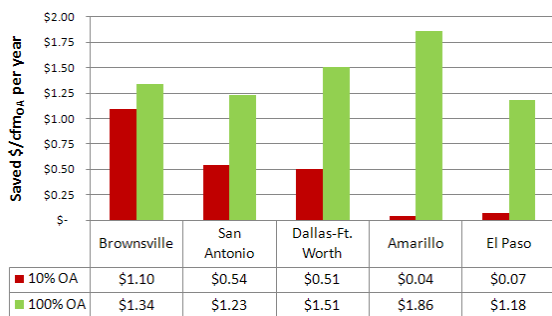


Figure 3: Projected Yearly Savings for an ERV at selected Outside Air Fractions.

An ERV manufacturer provided a rough cost estimate of \$2.75 per CFM of outside air for large, commercial ERVs. Figure 4 shows the payback time for various Texas cities at 10% and 100% Outside Air fractions using a \$2.75/ CFM_{OA} general ERV cost and the normalized savings from Figure 3. Note that payback periods for Amarillo and El Paso are long for low Outside Air fractions.

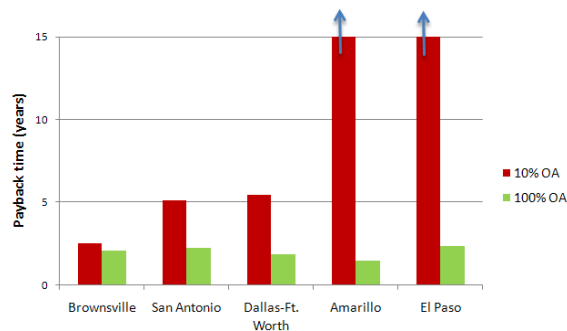


Figure 4: Payback time for ERV at selected Outside Air Fractions.

In order to better understand the reasons for the large differences in savings that is shown in Figure 3, it is useful to consider the components that make up the savings. Figure 5 shows the various component savings for San Antonio and Amarillo at 10% and 100% Outside Air fractions. The reported values are normalized savings (Saved $\$/\text{CFM}_{\text{OA}}$ per year) for each respective component. ERV Fan Energy is not a saving—but rather a cost—so it is shown as negative savings. The next component is Sensible Cooling; note that this component does not vary with the Outside Air fraction. San Antonio saves about twice the amount of Sensible Cooling as Amarillo. Savings for the ERV Latent Cooling is substantial for San Antonio but nonexistent for Amarillo. Savings for ERV Sensible Heating is substantial for 100% Outside Air fractions, but nonexistent for 10% Outside Air fractions. This is due to the ERV control logic for Scenario #3: for low Outside Air fractions (e.g. 10%), the Mixed Air temperature isn't affected very much by cold outdoor conditions (recall Table 1).

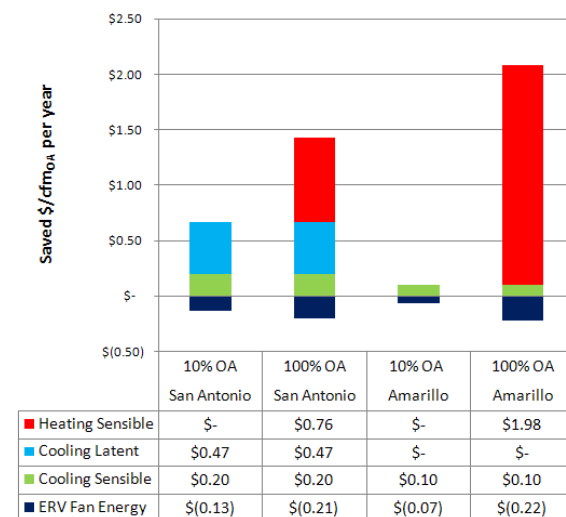


Figure 5: Savings by Energy Component for San Antonio and Amarillo.

Table 2: Predicted Annual Energy Savings from Application of an ERV to Various Buildings.

#	Building Type	Location	Area (ft ²)	OA Fraction	Without ERV Annual Energy Cost	With ERV		
						Saved \$/cfm _{OA} per year	Annual Energy Cost	
							Savings	% Savings
1	Office/Classroom	College Station	182,000	22%	\$ 647,378	0.62	\$ 15,161	2.3%
2	Office/Classroom	College Station	133,000	10%	\$ 206,789	0.39	\$ 6,629	3.2%
3	Office/Classroom	College Station	105,000	30%	\$ 285,566	0.39	\$ 9,468	3.3%
4	Laboratory	Houston	207,000	65%	\$ 1,844,012	0.88	\$ 164,380	8.9%
5	Laboratory	College Station	150,000	85%	\$ 797,452	0.80	\$ 97,194	12.2%

Potential Energy Cost Savings from Applying ERVs to Five Texas Buildings

Using internal reports created by the Energy Systems Laboratory, three office/classroom buildings and two laboratory buildings in Texas were analyzed to determine potential savings from an application of ERV technology to these buildings. For each building, an ERV analysis was conducted using the same inputs as specified in Appendix A, except for the following building parameters: Outside Airflow⁷ (CFM), AHU Airflow (CFM), Cooling Unit Cost (\$/MMBtu), Heating Unit Cost (\$/MMBtu), Electricity Unit Cost (\$/kWh), and Weather data⁸. These parameters were determined from the internal CC© reports.

Table 2 gives building descriptions and the predicted annual energy cost savings from applying ERV technology. The Annual Energy Cost is the sum of the chilled water cost, hot water cost, and total electricity cost for the whole building on an annual basis. The normalized total energy cost savings (Saved \$/CFM_{OA} per year) that are given do not include savings from ERV latent heating. For buildings with humidification units (such as some laboratories) the energy cost savings may be higher than listed.

CONCLUSION

This preliminary analysis has considered the ERV savings for various Outside Air fractions and locations in Texas. From Figure 3 it may be concluded that:

- For a typical laboratory building that requires 100% outside air, an ERV could save roughly \$1.00 to \$1.50 per cubic foot

per minute (CFM) of outside air during a one year period.

- For a typical office building that only requires 10% outside air, an ERV could save negligible savings up to \$1.00 per CFM of outside air over the period of one year.

A similar ERV savings analysis by the U.S. E.P.A.'s *Laboratories for the 21st Century* gives consistent results if the energy costs they used are calculated with current prices (Reilly & Van Geet 2003). Further comparison is difficult because of differences in analysis assumptions and their considered weather data did not include Texas.

Summarizing the results shown in Figure 4 gives:

- For a laboratory building in Texas or an office building in the Rio Grande Valley, payback time is less than 3 years.
- For an office building in other parts of Texas, payback time would range from 5 years to greater than the lifetime of the ERV.

In general, these payback periods are consistent with those reported by ASHRAE: "Well-designed energy recovery systems normally have a [payback period] of less than 5 years, and often less than 3 years. Paybacks of less than 1 year are not uncommon in comfort-to-comfort applications in hot, humid climates, primarily because of the reduced size of cooling equipment required." (ASHRAE Systems 2004)

Savings from ERV Latent Heating is not considered in this analysis; buildings with humidification systems will undoubtedly find additional savings and decreased payback times than reported here.

Due to the wide variation in savings and economic payback times found in this analysis, we do not recommend blanket installation of ERVs in all Texas commercial buildings. On the other hand, based on manufacturers' specifications, the cost savings for

⁷ Outside and AHU Airflow values are measured values, unless measurements are unavailable in which case maximum design values are used.

⁸ San Antonio weather data was used for the buildings in Houston and College Station because these cities were not included in the original WYEC weather files.

applying ERVs in some buildings are significant with attractive payback times and such installations should be pursued. It should be noted that this preliminary analysis has shown that savings are strongly dependent on the building type and location in Texas. Therefore, a comprehensive analysis that simulates all representative Texas commercial building air-handling systems in varying climates under typical operating conditions is required to obtain a complete understanding of the value of these systems.

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- ## APPENDIX A: ANALYSIS ASSUMPTIONS AND PARAMETER VALUES
- ### Analysis Assumptions
- Energy savings from ERV use is determined by comparing the energy required to condition “Outside Air to Return Air” conditions with the energy required to condition “ERV-supplied Fresh Air to Return Air” conditions.
- This analysis does not consider interactions with the air-handling system; it simply interfaces at the Mixed Air and Return Air streams (see Figure 1). Therefore the effects of economizers, variable flowrates, air distribution (single-duct vs. dual-duct), terminal reheat schemes, and other system components have not been examined here. We believe that these interactions with the overall air-handling system could significantly affect the cost effectiveness of an ERV.
- The assumptions for this analysis include:
- Overall System assumptions:
 - Outside Air provided 24 hours per day throughout the year at a constant flowrate.
 - Return Air from zones provided 24 hours per day throughout the year at a constant flowrate and psychrometric conditions are held constant.
 - No building humidification systems.
 - ERV assumptions:
 - Constant effectiveness ϵ regardless of the conditions of the airstreams.
 - No cross-leakage between the air streams.
 - No water vapor condensation.
 - Steady-state operating conditions.
 - Only water vapor passes through the ERV membrane.
 - Two draw-through fans with motors located in the air streams (one for each air stream).
 - ERV control:
 - A bypass around the ERV is opened if and only if the ERV is off.
 - Scenario #3 (see Table 1) is used to control the ERV. The ERV is considered to be ON if $T_{OA} > T_{RA}$ or $T_{MA,bypass\ open} < T_{ERV\ Cutoff}$.
 - Air Stream assumptions:
 - Fresh Air stream and Exhaust Air stream have identical:
 - specific heat
 - dry air mass flowrate
 - enthalpy of vaporization h_{fg}
 - Fresh Air stream, Recirculated Air stream, and Mixed Air stream have identical:

- specific heat
- density
- Cost assumptions:
 - Energy Costs are assumed to be flat-rate (no demand pricing).
 - Retrofit Costs were not considered.
 - Maintenance Costs were not considered.
 - NOTE: Anecdotal evidence suggests that maintenance costs may be significant.
 - Inflation, interest, and taxes were not considered in the analysis.

Parameter Values

The input values used in this analysis include

- ERV Effectiveness & Control:
 - Sensible effectiveness ε_s of 64%
 - Latent effectiveness ε_L of 60%
 - ERV Cutoff Temperature $T_{ERV\ cutoff}$ of 55°F
- ERV fan parameters:
 - 1.0 inch of water in pressure drop across the device (both on the Fresh Air stream and Exhaust stream,
 - fan & motor efficiencies for both fans are 70% and 85%, respectively.
- AHU parameters⁹:
 - AHU Airflow of 25,000 CFM
 - Return Air temperature of 75 °F and relative humidity of 50%
- Energy costs:
 - Cooling cost of \$10/MMBtu
 - Heating cost of \$20/MMBtu
 - Electricity cost of \$0.10/kWh
- Nominal Air Constants:
 - Air pressure of 14.696 psia
 - Air Sensible Heat Factor of 1.1 Btu/hr·°F·CFM
 - Air Latent Heat Factor of 4840 Btu/hr·CFM

Weather data was obtained from the ASHRAE Binread program, which gives weather data for the original WYEC cities in 5°F bins (Degelman 1994). The available Texas cities available are Brownsville, San Antonio, Dallas, Amarillo, and El Paso. Weather data in temperature bins consisted of

1. dry-bulb air temperature (in °F),
2. number of hours per year that fall within the bin, and
3. mean-coincident wet-bulb temperature (in °F) for the bin.

APPENDIX B: ANALYSIS DEVELOPMENT

⁹ A value for Outside Airflow is not given here because it was varied.

Sensible Heat Transfer

The sensible effectiveness of an ERV is given as (ASHRAE 2004, p. 44.2)

$$\varepsilon_s = \frac{q_s}{q_{s,max}} = \frac{m_1 c_{p,1} (T_2 - T_1)}{C_{min} (T_3 - T_1)} = \frac{m_1 c_{p,3} (T_3 - T_4)}{C_{min} (T_3 - T_1)} \quad (21)$$

where q_s is the actual sensible heat transfer rate given by

$$q_s = \varepsilon_s q_{s,max}$$

where the maximum sensible heat transfer rate is given by

$$q_{s,max} = 60 C_{min} (T_3 - T_1) \quad (22)$$

and where

ε_s = sensible effectiveness of the ERV,
 T_n = dry-bulb temperature at various locations n in Figure 1 (in °F units),
 m_1 = Outside Air dry air mass flow rate (in lb/min),
 m_3 = Exhaust Air dry air mass flow rate (in lb/min),
 C_{min} = minimum of $c_{p,1} m_1$ and $c_{p,3} m_3$,
 $c_{p,1}$ = Outside Air moist air specific heat at constant pressure (Btu/lb·F), and
 $c_{p,3}$ = Exhaust Air moist air specific heat at constant pressure (Btu/lb·F).

If we assume no water vapor condensation in the ERV, then solving Eq. (21) for the temperature T_2 of the Outside Air after exchanging sensible heat with the Exhaust Air gives

$$T_2 = T_1 - \varepsilon_s \frac{C_{min}}{m_1 c_{p,1}} (T_1 - T_3). \quad (23)$$

Eq. (21), (22), and (23) “assume steady-state operating conditions, no heat or moisture transfer between the heat exchanger and its surroundings, no cross-leakage, and no energy gains from motors, fans, or frost control devices. Furthermore, condensation or frosting does not occur or is negligible”.

If we further assume that the specific heat and dry air mass flow rate for both streams (Fresh Air and Exhaust Air) are identical, then Eq. (23) can be reduced to

$$T_2 = T_1 - \varepsilon_s(T_1 - T_3). \quad (24)$$

Latent Heat Transfer

ERVs not only transfer sensible heat, but they also transfer latent heat between the two air streams. If we assume no water condensation and that only water vapor is allowed to pass between the two air streams, then the latent effectiveness ε_L is given as (ASHRAE 2004, 44.3)

$$\begin{aligned} \varepsilon_L &= \frac{q_L}{q_{L,max}} = \frac{m_1 h_{fg} (W_1 - W_2)}{m_{min} h_{fg} (W_1 - W_3)} \\ &= \frac{m_3 h_{fg} (W_4 - W_3)}{m_{min} h_{fg} (W_1 - W_3)} \end{aligned} \quad (25)$$

where q_L is the actual latent heat transfer rate given by

$$q_L = \varepsilon_L q_{L,max} \quad (26)$$

and the maximum latent heat transfer rate given by

$$q_{L,max} = 60 m_{min} h_{fg} (W_1 - W_3). \quad (27)$$

In Eq. (25) through (27),

ε_L = latent effectiveness,
 h_{fg} = enthalpy of vaporization in Btu/lb units,
 W_n = humidity ratio at various locations n in Figure 1 (in lb. moisture/lb. dry-air), and
 m_{min} = minimum of m_1 and m_3 , which were both defined earlier.

If we assume constant enthalpy of vaporization h_{fg} and solve Eq. (25) for the humidity ratio W_2 of the Outside Air after exchanging latent heat with the Exhaust Air, then

$$W_2 = W_1 - \varepsilon_L \frac{m_{min}}{m_1} (W_1 - W_3). \quad (28)$$

Since we've already assumed that the dry air mass flowrates for both air streams are identical ($m_1 = m_3 = m_{min}$), then Eq. (28) further reduces to

$$W_2 = W_1 - \varepsilon_L (W_1 - W_3). \quad (29)$$

ERV Fans

In this analysis, we have assumed the ERV has its own supply and exhaust fan, with both operating in a draw-through fashion.

ERV Fan Energy Use.

The ERV fan motor energy use is either accounted

- Directly, by specifying the fan power required for each fan, or
- Indirectly, by specifying the air pressure drop across the fans, volumetric air flowrate passing through the fan, and a combined efficiency which is defined next.

If the indirect method is used, the fan power for each fan is calculated as (Kreider 2002, 216):

$$P_{fan} = \frac{\dot{V}_{OA} \Delta p}{6356 \eta_{combined}} \cdot \frac{1}{1.341} \quad (30)$$

where the fan power P_{fan} is in kiloWatts, the Outside Air flowrate \dot{V}_{OA} is in CFM, the pressure rise Δp across the fan is in inches of water, and the combined efficiency is defined as the product of the ERV fan motor efficiency and the ERV fan efficiency: $\eta_{combined} = \eta_{motor} \times \eta_{fan}$. The 1.341 factor is used to convert the power from hp to kiloWatts.

The total fan power consumed by the ERV is the sum of the fan power on the supply side and the fan power on the exhaust side of the ERV:

$$P_{fan,Total} = P_{fan,Supply} + P_{fan,Exhaust} \quad (31)$$

where the units are kiloWatts.

Temperature Rise from ERV Fan.

Next we account for the temperature rise of the air across the draw-through fan. For a control volume operating at a steady-state condition, negligible changes in kinetic and potential energies, and no work across the boundary, then from the First Law of Thermodynamics the heat transfer rate \dot{q} into or from a control volume depends on the mass flow rate \dot{m} and the entering and leaving enthalpies h (ASHRAE Fundamentals 2005, p. 1.2):

$$\dot{q} = \dot{m}(h_{out} - h_{in}). \quad (32)$$

If we further assume that the specific heat of the air is independent of temperature, then

$$\dot{q} = \dot{m} c_p (T_{out} - T_{in}), \quad (33)$$

where the air's specific heat is at a constant pressure c_p . Further adapting Eq. (33) for volumetric airflow:

$$\dot{q} = C_s \dot{V} (T_{out} - T_{in}), \quad (34)$$

where C_s (often taken as 1.08) is the air sensible heat factor in Btu/hr·°F·CFM units, the airflow \dot{V} is in CFM units, the temperature of the air entering and leaving the control volume is in Fahrenheit units, and heat transfer rate \dot{q} is in Btu/hr units.

Applying Eq. (34) to a control volume surrounding a fan, and solving for the temperature change of air entering and leaving the control volume yields:

$$\Delta T_{fan} = \frac{\dot{q}}{C_s \dot{V}}. \quad (35)$$

Converting the heat transfer rate to the work rate (using the conversion relation 1 Watt = 3.412 Btu/hr) yields:

$$\Delta T_{fan} = \frac{P_{fan}}{C_s \dot{V}} \frac{1}{3.412}, \quad (36)$$

where the temperature rise ΔT_{fan} is in degrees Fahrenheit, the fan power P_{fan} is in Watts, and the volumetric air flowrate \dot{V} is in CFM.

When one includes the temperature rise from the draw-through fan, the Supply Air leaving the ERV is

$$\begin{aligned} T_2' &= T_2 + \Delta T_{fan} \\ &= T_2 + \frac{P_{fan}}{C_s \dot{V}} \frac{1}{3.412} \end{aligned} \quad (37)$$

where T_2 was defined earlier in Eq. (24).

Since the ERV's fan does not affect the humidity ratio, then

$$W_2' = W_2, \quad (38)$$

where W_2 was defined earlier in Eq. (29).

Air Mixing

The Outside Air Fraction (X_{OA}) is defined as the ratio of the Outside Airflow rate to the AHU Airflow rate:

$$X_{OA} = \frac{\dot{V}_{OA}}{\dot{V}_{AHU}}. \quad (39)$$

where

- Outside Airflow (\dot{V}_{OA}) is the rate (in CFM) of Outside Air that is required by the air-handler unit. In this analysis, the rate is held constant, although it may bypass the ERV depending on system configuration and conditions.
- The AHU Airflow (\dot{V}_{AHU}) is the rate (in CFM) of air that is required by the constant-volume air-handler unit.

This analysis assumes that these Return Air properties (X_{OA} , \dot{V}_{OA} , and \dot{V}_{AHU}) are constant. The psychrometric properties of Recirculated Air are the same as Return Air (whose psychrometric properties has given in Appendix B).

As shown in Figure 1, the Fresh Air delivered by the ERV is mixed with the Recirculated Air that returns from the zones. If we assume constant specific heat for the different air streams, then it can be shown that the temperature T and humidity ratio W of the Mixed Air are (Kreider 2002, 144)

$$T_{MA} = \frac{\dot{m}_{FA,dry} T_{FA} + \dot{m}_{RecAir,dry} T_{RecAir}}{\dot{m}_{FA,dry} + \dot{m}_{RecAir,dry}} \quad (40)$$

and

$$W_{MA} = \frac{\dot{m}_{FA,dry} W_{FA} + \dot{m}_{RecAir,dry} W_{RecAir}}{\dot{m}_{FA,dry} + \dot{m}_{RecAir,dry}} \quad (41)$$

where the mass flowrates \dot{m} are for the dry air and the subscripts FA , $RecAir$, and MA refer to Fresh Air, Recirculated Air, and Mixed Air, respectively.

If we further assume a constant density for the different air streams, then Eq. (40) and (41) can be reduced to

$$T_{MA} = \frac{\dot{V}_{FA} T_{FA} + \dot{V}_{RecAir} T_{RecAir}}{\dot{V}_{FA} + \dot{V}_{RecAir}} \quad (42)$$

$$= X_{FA} T_{FA} + (1 - X_{FA}) T_{RecAir}$$

and

$$\begin{aligned}
 W_{MA} &= \frac{\dot{V}_{FA} W_{FA} + \dot{V}_{RecAir} W_{RecAir}}{\dot{V}_{FA} + \dot{V}_{RecAir}} \\
 &= X_{FA} W_{FA} + (1 - X_{FA}) W_{RecAir}
 \end{aligned} \tag{43}$$

where the volumetric Fresh Air fraction is

$$X_{FA} = \frac{\dot{V}_{FA}}{\dot{V}_{MA}}. \tag{44}$$

If the ERV is off (and therefore the ERV bypass is open) then mixing the Outside Air with the Recirculated Air results in

$$T_{MA,Bypass\ open} = X_{OA} T_{OA} + (1 - X_{OA}) T_{RecAir} \tag{45}$$

$$\begin{aligned}
 W_{MA,Bypass\ open} &= X_{OA} W_{OA} \\
 &\quad + (1 - X_{OA}) W_{RecAir} .
 \end{aligned} \tag{46}$$